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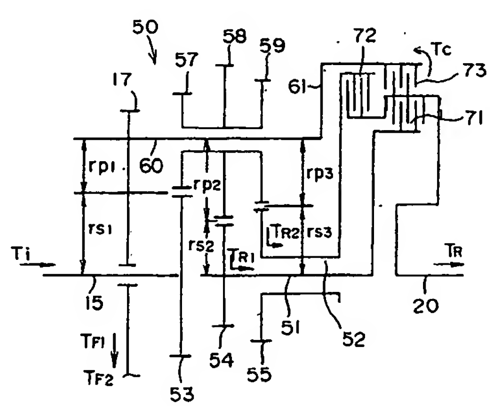
(54) **A system for controlling torque distribution in a four wheel drive vehicle.**

(57) A central differential for splitting output torque of a transmission is formed by a complex planetary device. The planetary device comprises an input sun gear operatively connected with an output shaft of the transmission, a plurality of output sun gears, a carrier, a pinion member comprising a plurality of planetary pinions integral with each other and rotatably supported on the carrier. Fluid operated friction clutches are provided for selectively connecting one of the output sun gears to an output member of the central differential.

FIG. 3



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A SYSTEM FOR CONTROLLING TORQUE DISTRIBUTION IN A FOUR WHEEL DRIVE VEHICLE

The present invention relates to a torque distribution control system for a four-wheel drive motor vehicle having a central differential, and more particularly to a control system which provides two or more standard torque distribution ratios.

In a four-wheel drive motor vehicle based on a front-wheel drive with a front-mounted engine which has a static weight ratio of 60 (front):40 (rear); the ratio of front torque T_F to the front wheels and a rear torque T_R to the rear wheels is determined to be 50:50 which is the same as the dynamic weight ratio. In a four-wheel drive motor vehicle based on a rear-wheel drive with a front-mounted engine which has a static weight ratio of 50:50, the ratio of the front torque T_F and the rear torque T_R is determined to be 40:60, the same dynamic weight ratio.

The front wheel drive based vehicle ensures safe driving on a slippery road. If a differential lock device is provided for locking the central differential, the driving force is improved. However, the steering of the vehicle is not improved. That is, when the vehicle makes a turn at high speed under the differential lock condition, all four wheels may slip (slipping spin) at the same time, causing difficulty in driving.

In order to ensure driving stability of the vehicle, the torque to the rear wheels is set to a larger value than that to the front wheels by arranging the central differential so that the rear wheels slip first. Thus, the vehicle can be safely driven by the front wheels at a small torque while a power transmission to the rear wheels causes idling.

Japanese Patent Application Laid-Open 63-176728 discloses a four-wheel drive motor vehicle in which a central differential comprising a simple planetary gear device is provided. An output of a transmission is transmitted to a carrier of the simple planetary gear device. The torque is distributed to the front wheels through either of a sun gear or a ring gear and to rear wheels through the other of the gears. The torque to the front and rear wheels is unequally distributed at a ratio determined by the difference between pitch circles of the sun gear and the ring gear. A fluid operated multiple-disk friction clutch is provided as a lock device for controlling the differential operation. A standard torque distribution ratio determined by the ratio of the pitch circles can not be changed, unless the diameters of the sun gear and the ring gear changes.

In order to increase the standard torque split ratio, the diameter of the sun gear is decreased or that of the ring gear is increased. However, in a power transmitting system where an axle differential for front or rear axles and the central differential are coaxially disposed, a plurality of shafts such as axles, a front drive shaft, an input shaft connected to the transmission and a rear drive shaft are disposed so as to penetrate the sun gear. Therefore, the diameter of the sun gear cannot be decreased. On the other hand, the ring gear cannot be large because of limitation of space and of required gear ratio. Consequently, the power transmitting system cannot be applied to various vehicles having different static weight ratio, such as a four wheel drive motor vehicle based on a front-wheel drive vehicle with a front-mounted engine, the four wheel drive motor vehicle based on a rear-wheel drive vehicle with a rear-mounted and mid-shipped engine.

Moreover, since the distribution of torque to the rear wheels can not be set to a large value, the control range of the multiple-disk friction clutch is small. Consequently, a motor vehicle having good drivability and steerability can not be provided.

It is desirable that the standard torque distribution ratio can be freely determined without dimensional restrictions so as to transmit a sufficient torque to the rear wheels. The drivability of the vehicle such as stability, cornering performance and steering feeling changes in dependency on whether the vehicle is driven in an understeering condition or in an oversteering condition. However, when only one standard torque distribution ratio is provided, only one of the conditions can be set so that it is difficult to comply with various road conditions and other driving conditions.

It is an object of the present invention to provide a system for controlling the torque distribution in a four wheel drive vehicle which alleviates the aforementioned technical problems experienced with the known prior art systems.

According to the present invention, there is provided a system for controling torque distribution in a four-wheel drive vehicle having a transmission and power trains for transmitting the torques to front and rear sets of drive wheels of the vehicle; comprising, a central differential having an input sun gear coupled to an output shaft of said transmission, a plurality of output sun gears coaxial with said input sun gear, a carrier coupled to one set of the driving wheels of said vehicle, a pinion member comprising a plurality of planetary pinions directly and coaxialy coupled to each other and each engaged with one of the sun gears to form an input planetary gear set and a plurality of output planetary gear sets, each of the output planetary gear sets having a drive ratio different to any other output planetary gear set, selecting means for engaging one of said output sun gears to an output member coupled to another set of the drive wheels, and a control unit for

operating said selecting means in accordance with the driving conditions of the vehicle.

In an aspect of the invention, the selecting means comprises fluid operated friction clutches.

A fluid operated multiple-disk clutch may be provided to restrict the differential operation of the central differential.

Other objects and features of this invention will be understood from the following description which refers to the accompanying drawings in which the word split may be used instead of distribution and:

Fig. 1 shows a schematic diagram of a power transmission system for a four-wheel drive motor vehicle of a first embodiment of the present invention;

Figs. 2a and 2b show an enlarged sectional view of a central differential and a fluid operated clutch device of the system;

Fig. 3 is a schematic view showing the central differential and the clutch device;

Figs. 4a and 4b show a circuit of a control system for a hydraulic clutches of the clutch device;

Fig. 5a is a graph showing a relationship between front torque and rear torque;

Fig. 5b is graph showing characteristics of a clutch pressure in accordance with a slip ratio;

Fig. 6 is schematic diagram showing the central differential, a clutch device, and a circuit for controlling the clutch device of a modification of the first embodiment;

Figs. 7a and 7b show a circuit of a control system for the hydraulic clutches of a second embodiment of the present invention;

Figs. 8a, 8b and 8c are graphs showing clutch pressures in first, second and third clutches of the clutch device in the control system of the second embodiment;

Fig. 9 is a schematic diagram showing a part of a circuit for controlling the clutches in the modification of the second embodiment;

Fig. 10 is a schematic view showing the central differential and the clutch device of a control system of a third embodiment of the present embodiment;

Figs. 11a and 11b show a circuit of a control system for a hydraulic clutches of the third embodiment;

Fig. 12a is a graph showing a relationship between front torque and rear torque in the third embodiment;

Fig. 12b is a graph showing characteristics of a clutch pressure in accordance with a slip ratio; and

Figs. 13a, 13b and 13c are explanatory illustrations showing engagement of pinions of a central differential provided in a fourth embodiment of the present invention.

Fig. 1 shows a power transmission system in a transaxle type for a four-wheel drive motor vehicle having an engine at a front portion thereof. The power transmission system has a torque converter 13 with a lockup clutch 12 mounted in a converter box 1, and a front differential 19 mounted in a differential box 2 behind the torque converter 13. A transmission case 3 housing an automatic transmission is attached to a rear of the differential box 2. An oil pan 5 is attached to an underside of the transmission case 3. A crankshaft 11 of the engine 10 is operatively connected with the torque converter 13. An input shaft 14 extends from a turbine of the torque converter 13 to the automatic transmission 30. Output of the automatic transmission 30 is transmitted to an output shaft 15 which is aligned with the input shaft 14 for rearwardly transmitting the torque. The output shaft 15 is connected to a front drive shaft 16 which is parallelly disposed under the automatic transmission 30 through a pair of reduction gears 17 and 18 of a central differential 50 housed in a transfer case 4. A fluid operated clutch device 70 housed in an intermediate case 6 is disposed behind the central differential 50. The front drive shaft 16 is connected to front wheels through a front differential 19. The output shaft 15 is connected to a rear drive shaft 20 provided in an extension case 7 through the central differential 50 and the clutch device 70. The rear drive shaft 20 is connected to rear wheels through a propeller shaft 21 and a rear differential 22. The automatic transmission 30 comprises two sets of single planetary gears consisting of a front planetary gear 31 and a rear planetary gear 32 for providing four forward speeds and one reverse speed. The automatic transmission has a high clutch 33, a reverse clutch 34, a brake band 35, a forward clutch 36, an overrunning clutch 37, a low and reverse clutch 38, and one-way clutches 39 and 40.

An oil pump 41 is provided in a front end of the transmission case 3. A pump drive shaft 42 is connected to an impeller 13a of the torque converter 13 and is operatively connected with a rotor of the oil pump 41.

In the oil pan 5, a control valve body 43 is provided for hydraulically actuating respective clutches and a brake of the transmission 30.

Referring to Figs. 2a and 2b showing the central differential 50 and the clutch device 70, a first intermediate shaft 51 is rotatably mounted in the output shaft 15 at a front portion thereof through bushes 23 and a thrust washer 24. A rear portion of the intermediate shaft 51 is rotatably mounted in the rear driveshaft 20. A second intermediate shaft 52 is rotatably mounted on the rear portion of the first intermediate shaft 51 through bushes 23a. The reduction gear 17 is rotatably mounted on the output shaft

15 through a bush 23b and a thrust bearing 25. The reduction gear 17 and the output shaft 15 are mounted in the transmission case 3 through a ball bearing 26.

The central differential 50 is a complex planetary gear device and comprises a first sun gear 53 formed on the output shaft 15, a first planetary pinion 57 meshed with the first sun gear 53, a second sun gear 54 formed on the first intermediate shaft 51, a second planetary pinion 58 meshed with the second sun gear 54, a third sun gear 55 formed on the second intermediate shaft 52, a third planetary pinion 59 meshed with the third sun gear 55, and a carrier 61 secured to the reduction gear 17 through connecting members 60. The first to third planetary pinions 57 to 59 are integral with one another to form a pinion member 56. The pinion member 56 is rotatably mounted on a shaft 62 through needle bearings 28. The shaft 62 is secured to the gear 17 and the carrier 61. A boss 61a of the carrier 61 formed at the rear end thereof is rotatably mounted on the second intermediate shaft 52 through a bearing 27. The carrier 61 is rotatably supported in the transfer case 4 through a ball bearing 26a.

The fluid operated clutch device 70 in the intermediate case 6 comprises first and second fluid operated multiple disk friction clutches 71 and 72 for changing the standard torque split ratios to the front and the rear wheels, and a third fluid operated multiple-disk friction clutch 73 for restricting the operation of the central differential 50. The first clutch 71 is provided between the first intermediate shaft 51 and the rear drive shaft 20, and comprises a driven drum 71a splined on a hub 20a of the rear drive shaft 20 and rotatably mounted on a boss 6a of the intermediate case 6 formed at the inner portion thereof, and a drive drum 71b splined on the first intermediate shaft 51. A plurality of driven disks 71c are splined on the driven drum 71a and a plurality of drive disks 71c′ are splined on the drive drum 71b, disposed alternately with the disks 71c. A ring piston 71e is slidably mounted on the inner wall of the driven drum 71a. The piston 71e engages with the end disks 71c. An oil chamber 71d is defined between the piston 71e and the driven drum 71a. When the oil is supplied to the chamber 71d, the piston 71e is pushed by the pressure of the oil. Thus, the disks 71c and 71c′ are engaged with the adjacent disks to engage the clutch 71.

Thus, the output torque from the output shaft 15 of the transmission 30 is transmitted to the carrier 61 and the second sun gear 54 through the first sun gear 53 and the pinions 57, 58 at predetermined respective torque split ratios. The difference between rotating speeds of the carrier 61 and the second sun gear 54 is absorbed by the rotation of the first and second planetary pinions 57 and 58. The torque is further transmitted to the rear drive shaft 20 through the first intermediate shaft 51 and the first clutch 71.

The second clutch 72 comprises a drive drum 72a secured to the second intermediate shaft 52 and a driven drum 72b securely mounted on an inner wall of the driven drum 71a of the first clutch 71. A plurality of drive disks 72c are splined on the drive drum 72a and a plurality of driven disks 72c′ are splined on the driven drum 72b, disposed alternately with the disks 72c. A ring piston 72e is slidably mounted on the second intermediate shaft 52. The piston 72e engages with the end disk 72c. An oil chamber 72d is defined between the piston 72e and the drive drum 72a.

When the oil is supplied to the oil chamber 72d, the disks 72c and 7c′ are engaged to engage the clutch 72, thereby coupling the second intermediate shaft 52 with the rear drive shaft 20 through driven drums 72b and 71a. Hence the output torque from the output shaft 15 of the transmission 30 is transmitted to the third sun gear 55 through the first sun gear 53 and the pinions 57 and 59 at predetermined respective torque split ratios. The difference between the rotating speeds of the carrier 61 and the third sun gear 55 is absorbed by the rotation of the first and third planetary pinions 57 and 59. The torque is further transmitted to the rear drive shaft 20 through the second intermediate shaft 52 and the second clutch 72 and the driven drum 71a of the first clutch 71. Thus, the first sun gear 53 acts as an input member, and the second and third sun gears act as output members.

The third clutch 73 has a drum 73a which is secured on the boss 61a of the carrier 61 of the central differential 50, and surrounds the driven drum 71a of the first clutch 71. A plurality of disks 73c splined on the drum 73a are alternately arranged with a plurality of disks 73c′ which are splined on the outer periphery of the driven drum 71a of the first clutch 71. A piston 73e is slidably mounted on an inner wall 6b of the inter mediate case 6. A retainer 73g mounted on the piston 73e through a release bearing 73f is abutted on the innermost disk 73c. An oil chamber 73d is formed between the piston 73e and the intermediate case 6.

When oil is supplied to the chamber 73d, the piston 73e is pushed by the pressure of the oil. The piston 73e pushes the retainer 73g which in turn pushes the disks 73c and 73c′ to engage the clutch 73 for producing a clutch torque.

A pair of thrust bearings 25a are provided at both ends of the second intermediate shaft 52. Thrust bearings 25b are provided between the drive drum 71b and the flange 20a of the rear drive shaft 20 and between the flange 20a and the boss 6a of the intermediate case 6. Thus, each element can be freely rotated.

Oil passages 15a and 51a are formed in the output shaft and the first intermediate shaft 51,

respectively. Oil ports 51b and 51c are formed in the intermediate shaft 51. The lubricating oil is fed to the passage 51a from an oil passage 3a formed in the transfer case 3 through passages 15b and 15a, and further fed to bushes 23 for lubricating the bushes, the first sun gear 53 and pinions 57. An oil port 51c communicated with the passage 51a is provided for lubricating the bush 23a, thrust bearing 25a, second and third sun gears 54, 55 and pinions 58, 59.

An oil passage 78 is formed in the boss 6a of the intermediate case 6 for supplying oil to the oil chamber 71d of the first clutch 71. The oil chamber 72d of the second clutch 72 is supplied with oil through an oil passage 79 formed in the intermediate case 6, rear drive shaft 20 and the first immediate shaft 51.

The operation of the central differential 50 for splitting the torque to the front and rear wheels will be described hereinafter with reference to Fig. 3.

An input torque Ti of the first sun gear 53 and the relationship between the sun gears and the pinions are expressed as follows, respectively.

$Ti = T_{F1} + T_{R1}$ (1)

$Ti = T_{F2} + T_{R2}$ (2)

$rs_1 + rp_1$
$= rs_3 + rp_2$
$= rs_3 + rp_3$ (3)

where T_{F1} and T_{F2} are the front torques transmitted from the carrier 61 to the front drive shaft 16, T_{R1} is the rear torque transmitted from the second sun gear 54 to the rear drive shaft 20, T_{R2} is the rear torque transmitted from the third sun gear 55 to the rear drive shaft 20, $rs3_1$ is the radius of the pitch circle of the first sun gear 53, rp_1, rp_2 and rp_3 are radii of pitch circles of first, second and third pinions 57, 58 and 59, respectively, and rs_2 and rs_3 are the radii of the pitch circles of the second and the third sun gears 54 and 55.

A tangential load P on the engaging point of the first sun gear 53 and the first pinion 57 is equal to the sum of a tangential load P_1 on the carrier 61 and a tangential load P_2 on the engaging point of the second sun gear 54 and the second pinion 58. The tangential load P is further equal to the sum of a tangential load P_3 on the carrier 61 and a tangential load P_4 on the engaging point of the third sun gear 55 and the third pinion 59. That is,

$P = Ti/rs_1$

$P_1 = T_{F1}(rs_1 + rp_1)$

$P_2 = T_{R1}/rs_2$

$P_3 = T_{F2}/(rs_1 + rp_1)$

$P_4 = T_{R2}/(rs_3 + rp_3)$

$Ti/rs_1 = \{ T_F/(rs_1 + rp_{1)} \} + T_R/rs_2$ (4)

$Ti/rs_1 = \{ T_{F2}/(rs_1 + rp_{1)} \} + T_{R2}/rs_3$ (5)

Substituting equations (1) to (3) for the equation (4) and (5),

$T_{F1} = (1 - rp_1 \cdot rs_2/rs_1 \cdot rp_2) \cdot Ti$ (6)

$T_{R1} = (rp_1 \cdot rs_2/rs_1 \cdot rp_2) \cdot Ti$ (7)

$T_{F2} = (1 - rp_1 \cdot rs_3/rs_1 \cdot rp_3) \cdot Ti$ (8)

$T_{R2} = (rp_1 \cdot rs_3/rs_1 \cdot rp_3) \cdot Ti$ (9)

Consequently, it will be seen that the torque split for the front torque T_F and the rear torque T_R can be set to various values by changing the radii of the pitch circles of the sun gears 53 to 55 and the pinions 57 to 59. Furthermore, since three sets of sun gear and pinions are provided, two standard torque split ratios are obtained.

If rs_1 is 22.8 mm, rp_1 is 17.1 mm, rp_2 is 21.8 mm, rs_2 is 18.1 mm, rp_3 is 19.95 mm, and rs_3 is 19.95 mm, the front torque T_F and the rear torque T_R are calculated as

$T_{F1} = 0.38Ti$

$T_{R1} = 0.62Ti$

$T_{F2} = 0.25Ti$

$T_{R2} = 0.75Ti$

Thus, the standard torque split ratios of the front wheels and the rear wheels are

$T_{F1} : T_{R1} \fallingdotseq 38 : 62$

$T_{F2} : T_{R2} \fallingdotseq 25 : 75$

A large torque can be distributed to the rear wheels, and particularly at the second torque split ratio, a larger torque is distributed to the rear wheels as in the rear-drive vehicle.

Referring to Figs. 4a and 4b showing a control system for the clutches 71, 72, 73, the oil pressure control unit of the control system comprises a pressure regulator valve 75, a pilot valve 83, a clutch control valve 81 and a solenoid operated duty control valve 84 for controlling the third fluid operated multiple-disk

clutch 73. The regulator valve 75 operates to regulate the pressure of oil supplied from the oil pump 41 driven by the engine 10 to produce a predetermined line pressure and a lubricating oil pressure. An actuating pressure conduit 76 is communicated with a passage 85 through the pilot valve 83. The passage 85 is communicated with the solenoid operated duty control valve 84 and with an end port of the clutch control valve 81. The conduit 76 is communicated with the clutch control valve 81 through a passage 76a. The clutch control valve 81 is communicated with the third clutch 73 through the passage 82. The solenoid operated valve 84 is operated by pulses from a control unit 90 at a duty ratio determined therein, so as to control draining the oil to provide a control pressure. The control pressure is applied to an end of a spool of the clutch control valve 81 to control the oil supplied to the clutch 73 so as to control the clutch pressure (torque). The passage 76a is further communicated with a changeover valve 77 through a passage 76b. The changeover valve 77 has a solenoid 77a which is energized when a manual switch 80 connected thereto is closed. The changeover valve 77 is thus operated to selectively communicate the passage 76a with the oil chamber 71d of the first clutch 71 through a passage 78 or the oil chamber 72d of the second clutch 72 through a passage 79.

The control unit 90 is supplied with output signal from a front-wheel speed sensor 91, a rear-wheel speed sensor 92 and a steering angle sensor 93.

The control unit 90 has a slip ratio calculator 94 to which the front-wheel and rear-wheel speeds N_F and N_R are applied. Since the standard torque split is determined in accordance with the principle of $T_F < T_R$, the rear wheels slip first (slipping spin). A slip ratio S is calculated in accordance with the ratio of the front-wheel speed N_F to the rear-wheel speed N_R, $S = N_F/N_R (S > 0)$. The slip ratio S, and a steering angle ψ from the sensor 93 are applied to a clutch pressure setting section 95. In accordance with the input signals, the clutch pressure setting section 95 retrieves a clutch pressure Pc from a clutch pressure look up table 96. When the slip ratio S is:$S \geqq 1$, the clutch pressure Pc is set to a small value. When the rear wheels slip and the slip ratio S becomes:$S < 1$, the clutch pressure Pc (clutch torque) increases with a decrease of the slip ratio S. When the slip ratio S becomes smaller than a set value Sl, the clutch pressure Pc is set to a maximum Pcmax. Further, when the steering angle ψ increases, the clutch pressure Pc is decreased, thereby preventing the tight corner braking.

The clutch pressure Pc is applied to a duty ratio providing section 97 where a duty ratio D corresponding to the derived clutch pressure Pc is provided. A duty signal with a duty ratio D provided at the section 97 is applied to the solenoid operated duty control valve 84.

Describing the operation of the system with reference to Figs. 5a and 5b, the power of the engine 10 is transmitted through the torque converter 13 and the input shaft 14 to the transmission 30 at which the transmission ratio is automatically controlled. The output of the transmission 30 is transmitted to the first sun gear 53 of the central differential 50.

In order to stably drive the vehicle on roads in general, the manual switch 80 is opened so that the changeover valve 77 is operated to communicate the passage 76b with the passage 78. The first clutch 71 is engaged, thereby to connect the second sun gear 54 of the central differential 50 with the rear drive shaft 20 through the first intermediate shaft 51 and the clutch 71. Thus, the first mode, wherein the torque is distributed in accordance with the first standard torque split ratio is selected. Namely, the standard torque split ratio is $T_{F1} : T_{R1} \fallingdotseq 38:62$, dependent on the radii of the first and second sun gears 53, 54 and the pinions 57, 58. Thus, 38% of the output torque of the transmission 30 is transmitted to the front wheels through the carrier 61, the reduction gears 17, 18, the first drive shaft 16 and the front differential 19. Meanwhile, 68% of the torque is transmitted to the rear wheels through the second sun gear 54, first intermediate shaft 51, first clutch 71, the rear drive shaft 20, the propeller shaft 21 and the rear differential 22. Thus, the four-wheel driving is established.

A slip ratio S is calculated in accordance with the front wheel speed N_F, the rear-wheel speed N_R and the steering angle ψ. If no slip state is detected in the control unit 90 while the vehicle is driven on the dry road ($S \geqq 1$), a low clutch pressure Pc is set in the clutch pressure setting section 95 so that a signal corresponding to the duty ratio of 100% is applied from the duty ratio providing section 97 to the solenoid operated duty control valve 84. Thus, the clutch control pressure becomes zero and the clutch control valve 81 operates to close the passage 76a, thereby draining the oil from the third clutch 73. The clutch 73 is disengaged and the clutch torque becomes zero so as to render the central differential 50 free.

Accordingly, the torque split to the front and rear wheels becomes the same as the standard split ratio $T_{F1} : T_{R1}$ as shown at a point P_1 of the graph of Fig. 5a.

At the first standard torque split ratio, the vehicle is driven under the understeering condition, so that good operability of the vehicle is ensured. Further, the vehicle smoothly negotiates a sharp corner owing to the differential operation of the central differential 50.

If the vehicle is driven on a slippery road, the rear wheels slip first because the larger amount of the

torque is distributed to the rear wheels. The slip ratio S_1 is calculated at the slip ratio calculator 94 of the control unit 90. A duty signal corresponding to a clutch pressure Pc_1 in accordance with the slip ratio S_1 (S <1) is applied to the solenoid operated valve 84. The clutch control valve 81 is operated by the control pressure of oil obtained by regulating the line pressure at the solenoid operated valve 84, so that the third clutch 73 is engaged at the clutch pressure. Consequently, the clutch torque Tc is produced in the clutch 73. The clutch 73 is provided in parallel with the carrier 61 and the second sun gear 54 of the central differential 50. Accordingly, the clutch torque Tc_1 corresponding to the slip ratio S_1 is transmitted from the second sun gear 54 to the carrier 61 to increase the torque to the front wheels. Thus, the split ratio of the front torque and the rear torque changes along a line l, as shown in Figs. 5a and 5b. To the contrary, the torque to the rear wheels is reduced to eliminate slipping, thereby improving driveability to ensure good operability and safe driving.

When the slip ratio S becomes smaller than the set value S_1, the differential operation restricting torque becomes maximum by the pressure of oil in the third clutch 73. Thus, the carrier 61 is directly engaged with the second sun gear 54 to lock the central differential 50. Thus, the four-wheel driving is established in accordance with the torque split corresponding to the axle loads of the front and rear wheels as shown at a point P_2. Thus, the torque split is continuously controlled in accordance with the slip condition for preventing the slipping of the four wheels.

The manual switch 80 is closed when the vehicle is driven on a mountainous road. As a result, the solenoid 77a of the changeover valve 77 is energized, thereby operating the valve 77 to communicate the passage 76b with the passage 79. Thus, the oil is supplied to the oil chamber 72d of the second clutch 72 to engage the clutch 72 so that the third sun gear 55 is connected to the rear drive shaft 20 through the second intermediate shaft 52, the second clutch 72, and the driven drum 71a of the first clutch 71. Accordingly, the second mode wherein the torque is distributed in accordance with the second standard torque split ratio is selected. That is, split ratio $T_{F2}:T_{R2}$ is 25:75 in accordance with the radii of the third sun gear 55 and the third pinions 59. The torque is mainly transmitted to the rear wheels as shown by a point P_1'. The vehicle is in an oversteering condition so that good cornering maneuverability, driveability and steerability are obtained.

Although the rear wheels tend to slip in the second mode, the clutch torque Tc is produced in the third clutch 73 when the slip occurs. The clutch torque Tc transmitted to the front wheels increases along a line l' shown in Fig. 5a. Thus, the split ratio changes between the points P_1' and P_2 at which the torque is equally distributed to the four wheels. Hence the slipping of the rear wheels is restrained to obtain good running performance.

Fig. 6 shows a modification of the embodiment of the torque split system. In the modification, the second sun gear 54 and the third sun gear 55 are connected to the rear drive shaft 20 through the first clutch 71 and the second clutch 72, resepctively. The system is provided with a control unit 100 which applies control signals to a first clutch control valve 77b communicated with the first clutch 71, and to a second clutch control valve 77c communicated with the second clutch 72. The control unit 100 is connected to the first, second and third switches 80a, 80b and 80c. When the first switch 80a is closed, the first clutch 71 is engaged to provided the first mode and when the second switch is closed, the second clutch 72 is engaged to provide the second mode. When the third switch 80c is closed, both clutches 71, 72 are engaged, thereby locking the central differential 50 to provide a four-wheel drive mode.

Since a third clutch for restricting the differential operation of the central differential 50 is not provided, it is preferable to construct the central differential 50 so as to automatically generate a restricting torque in proportion to the input torque, thereby restricting the differential operation.

In the torque split control system of the present invention, the first and second clutches serve as a power train to the rear wheels. At the time when one of the clutches is being drained whereas the other is being supplied with oil, if oil pressure is low, an accident that neither of the clutches are engaged occurs owing to low oil pressure, which means the cutting of the power train. If the front wheels or the rear wheels are on a road surface having a low friction coefficient in such a state and if the accelerator pedal is largely depressed, the torque is largely decreased and the engine speed abnormally increases.

In order to avoid such a decrease of torque, in the torque split control system of the second embodiment the present invention is provided with a hydraulic control system shown in Figs. 7a and 7b. A changeover signal from the switch 80 is applied to the clutch pressure setting section 95 of the control unit 90 when the torques distribution ratio mode is changed. Thus, the central differential 50 is prevented from becoming neutral.

More particuarly, when the switch 80 is closed to select the second torque split ratio mode, the oil in the first clutch 71 is drained so that the clutch pressure Pc_1 in the first clutch 71 starts to decrease as shown in the graph of Fig. 8a. The oil is supplied to the second clutch 72 so that the clutch pressure Pc_2 in

the second clutch 72 starts to increase as shown in the graph of Fig. 8b. At the same time the changeover signal is fed to a clutch pressure setting section 95 of the control unit 90. The clutch pressure setting section 95 provides a high clutch pressure Pc for a predetermined period t. As a result, a predetermined clutch torque Tc corresponding to the set clutch pressure Pc is generated in the third clutch 73, thereby temporarily rendering the central differential 50 in a locked state or a slipping state. The power train is maintained for the period t, in which time, the second clutch 72 is sufficiently engaged. After the period t, the clutch pressure Pc decreases thereby releasing the third clutch 73. The control system is operated in the same manner when the switch 80 is opened to change the torque split ratio mode from the second mode to the first mode.

Referring to Fig. 9 showing a hydraulic circuit of the control system of a modification of the second embodiment of the present invention, the releasing timings of the clutches are automatically retarded. The oil passage 78 connecting the changeover valve 77 with the first clutch 71 has a check valve 101a and an orifice 101b disposed in parallel with the check valve 101a. The oil passage 79 connecting the changeover valve 77 with the second clutch 72 similarly has a check valve 102a and an orifice 102b in pararell to the check valve 102a. When the changeover valve 77 is operated to connect the oil passage 76b with the passage 79, the oil is applied to the second clutch 72 passing the check valve 102a. To the contrary, the check valve 101a in the passage 78 is closed so that the oil is slowly drained through the orifice 101b. Hence as shown by a dotted line in Fig. 8a, the clutch pressure Pc_1 in the first clutch 71 gradually reduces so that the clutches are maintained to provide the first mode until the second clutch 72 is fully engaged.

Thus, in accordance with the second embodiment of the present invention, the central differential 50 is operated while the torque split ratios are changed so that the decrease of torque and the abnormal rising of the engine speed are prevented. In particular, the system where the differential 50 is locked by the third clutch, restrains a shock which occurs at the change of torque split ratio.

In the third embodiment of the present invention shown in Fig. 10, the central differential 50 is so arranged that the first sun gear 53 is equal to the third sun gear 55 and the first pinion 57 is equal to the third pinion 59 in the number of the teeth, the module, and the pitch circle. Therefore,

$rs_1 = rs_3, rp_1 = rp_3$

Substituting the above equations for the equations (6) to (9) hereinbefore described,

$T_{F1} = (1 - rp_1 \cdot rs_2/rs_1 \cdot rp_2) \cdot Ti$

$T_{R1} = (rp_1 \cdot rs_2/rs_1 \cdot rp_2) \cdot Ti$

$T_{F2} = 0, T_{R2} = Ti$

Thus, a second standard torque split ratio where the torque is completely split to the rear wheels is obtained.

If rs_1 and rs_3 are 22.8 mm, rp_1 and rp_3 are 17.1 mm, rp_2 is 21.8 mm and rs_2 is 18.1 mm, the front torque T_F and the rear torque T_R are calculated as

$T_{F1} \fallingdotseq 0.38Ti$

$T_{R1} \fallingdotseq 0.62Ti$

Thus, the first standard torque split ratio obtained through the second sun gear 54 is

$T_{F1} : T_{R1} \fallingdotseq 38:62$

Referring to Figs. 11a and 11b, the control unit 90 of the present embodiment is provided with a changeover determining section 98 which is fed with the slip ratio S from the slip ratio calculator 94. The changeover determining section 98 applies a signal to the solenoid 77a of the changeover valve 77 in dependency on the slip ratio S. Namely, while the wheels do not slip so that the slip ratio S is larger than a predetermined reference ratio S_R, the changeover valve 77 is operated to engage the second clutch 72. On the other hand, when the slip ratio S becomes smaller than the reference ratio S_R, the changeover valve 77 is automatically operated to engage the first clutch 71.

When the vehicle is driven on a dry road where the slipping of the wheels do not occur, the changeover determining means 98 feeds a signal to operate the solenoid 77a of the changeover valve 77 to communicate the oil passage 76b with the passage 79. The second clutch 72 is engaged so that the rear drive shaft 20 is connected to the third sun gear 55 through the second intermediate shaft 52. The second standard torque split ratio is selected as shown by a point P_3 in Fig. 12a, thereby transmitting the torque only to the rear wheels through the third sun gear 55, second intermediate shaft 52, second clutch 72, the rear drive shaft 20, the propeller shaft 21 and the rear differential 22. Hence, the vehicle is driven by the rear wheels although the central differential 50 is provided.

Meanwhile, the carrier 61 connected to the front wheels idles so that when the vehicle makes a turn, the carrier 61 is freely rotated. Since the vehicle is driven under the oversteering condition, maneuverability, driveability, and steerability are improved.

When the rear wheels slip so that the slip ratio S becomes smaller than the reference slip ratio S_R, the

changeover determining section 98 applies a signal to de-energize the solenoid 77a of the changeover valve 77, thereby communicating the passages 76b and 78. Thus, the first clutch 71 is engaged so that the rear drive shaft 20 is connected with the second sun gear 54 through the first intermediate shaft 51. The first standard torque split ratio mode is hence determined, thereby transmitting the torque to the front and the rear wheels at the ratio of $T_{F1}:T_{R1} \fallingdotseq 38:62$ as shown at the point P_1 in Fig. 12a. In the first standard torque split ratio mode, since the torque is distributed to the four wheels, slipping of the wheels is restrained, thereby enhancing good running performance and stable driving.

If an excessive torque is transmitted when driven on a slippery road, the rear wheels to which a larger torque is transmitted are apt to slip so that the slip ratio S decreases. The oil is applied to the third clutch 73 so that the clutch 73 is operated to increase the torque transmitted to the front wheels in the manner described in the description of the first embodiment. Therefore, the slipping of the wheels is prevented, and the vehicle is driven in an understeering condition so that the running performance and the stability of the vehicle are improved.

When the vehicle is driven on a road having an extremely low friction coefficient such as an icy road, or when stuck in mud or in sands, the slip ratio S becomes smaller than the set value S_1, the clutch pressure Pc applied to the third clutch 73 becomes maximum, thereby locking the central differential 50. The four-wheel driving is established so that the running performance is enhanced.

The present embodiment may be modified to manually operate the changeover valve, or in accordance with other factors besides the slip ratio S. In a vehicle having an antilock braking system on a brake system, when the antilock braking is effected, an ABS actuating signal is applied to the clutch pressure setting section 95 to render the clutch pressure Pc zero, thereby releasing the clutch 73.

In the central differential 50 of the fourth embodiment of the present invention, each set of the first to third pinion has three pinions.

The engagement conditions for equiangularly disposing a plurality of pinions will be described with reference to Figs. 13a to 13c.

Referring to Fig. 13a, if the first sun gear 53 is fixed and the first, second and third pinions 57, 58 and 59 are revolved an angle θ in the clockwise direction from a standard line SL, the second sun gear 54 is rotated by an angle α_1 in a counterclockwise direction and the third sun gear is rotated by an angle β_1 in a counterclockwise direction. The angle θ is expressed as follows.

$\theta = rs_2 \cdot rp_1 \cdot \alpha_1/(rs_1 \cdot rp_3 - rs_3 \cdot rp_1)$ (10)

$\theta = rs_3 \cdot rp_1 \cdot \beta_1/(rs_1 \cdot rp_2 - rs_2 \cdot rp_1)$ (11)

If the number of the teeth of the first sun gear 53 is Zs_1, the number of the teeth of the second sun gear 54 is Zs_2, the number of the teeth of the third sun gear is Zs_3, the number of the teeth of the first pinion 57 is Zp_1, the number of the teeth of the second pinion 58 is Zp_2, and the number of the teeth of the third pinion 59 is Zp_3, the equations (10) and (11) are substituted as follows in accordance with the numbers of the teeth.

$\theta = Zs_2 \cdot Zp_1 \cdot \alpha_1/(Zs_1 \cdot Zp_3 - Zs_3 \cdot Zp_1)$ (12)

$\theta = Zs_3 \cdot Zp_1 \cdot \beta_1/(Zs_1 \cdot Zp_2 - Zs_2 \cdot Zp_1)$ (13)

If the second sun gear 54 is rotated by a circular pitch angle $360/Zs_2$ and the third sun gear 55 is rotated by a circular angle $360/Zs_3$, each from the standard line, the angle θ is represented as

$\theta = Zp_1 \cdot 360(Zs_1 \cdot Zp_3 - Zs_3 \cdot Zp_1)$ (14)

$\theta = Zp_1 \cdot 360/(Zs_1 \cdot Zp_2 - Zs_2 \cdot Zp_1)$ (15)

Referring to Fig. 13b, if the second sun gear 54 is fixed and the first, second and third pinions 57, 58 and 59 are revolved by the angle θ in the clockwise direction, the first sun gear 53 is rotated by an angle β_2 in the clockwise direction and the third sun gear 55 is rotated by an angle α_2 in the clockwise direction. The angle θ is expressed as

$\theta = - Zs_3 \cdot Zp_2 \cdot \beta_2/(Zs_2 \cdot Zp_1 - Zs_1 \cdot Zp_2)$ (16)

$\theta = - Zs_1 \cdot Zp_2 \cdot \alpha_2/(Zs_3 \cdot Zp_2 - Zs_2 - Zs_2 \cdot Zp_3)$ (17)

Substituting $\beta_2 = 360/Zs_3$ and $\alpha_2 = 360/Zs_1$ in the equations (16) and (17),

$\theta = - Zp_2 \cdot 360/(Zs_2 \cdot Zp_1 - Zs_1 \cdot Zp_2)$ (18)

$\theta = - Zp_2 \cdot 360/(Zs_3 \cdot Zp_2 - Zs_2 \cdot Zp_3)$ (19)

Referring to Fig. 13c, if the third sun gear 55 is fixed and the first, second and third pinions 57, 58 and 59 are revolved by the angle θ in the clockwise direction, the first sun gear 53 is rotated by an angle β_3 in the clockwise direction and the second sun gear 54 is rotated by an angle α_3 in the clockwise direction. The angle θ is expressed as

$\theta = Zp_3 \cdot Zs_2 \cdot \alpha_3/(Zp_3 \cdot Zs_2 - Zp_2 \cdot Zs_3)$ (20)

$\theta = - Zp_3 \cdot Zs_1 \cdot \beta_3/(Zp_1 \cdot Zs_3 - Zs_1 \cdot Zp_3)$ (21)

Substituting $\alpha_3 = 360/Zs_2$ and $\beta_3 = 360/Zs_3$ in the equations (20) and (21). $\theta = Zp_3 \cdot 360/Zp_3 \cdot Zs_2$

$- Zp_2 \cdot Za_3)$ (22)

$\theta = - Zp_3 \cdot 360/(Zp_1 \cdot Zs_3 - Zs_1 \cdot Zp_3)$ (23)

If a number N of the pinons are equiangularly disposed, the disposition angle between a first pair of the pinions and a second pair of the pinions is 360/N. It will be seen that integer times of a value is obtained by integrally multiplied the number N from the equations (14), (15), (18), (19), (22) and (23). Consequently, for evenly spaced pinions the equation is

$m_1 = (Zs_1 \cdot Zp_3 - Zs_3 \cdot Zp_1)/Zp_1 \cdot N$

$= (Zs_1 \cdot Zp_2 - Zs_2 \cdot Zp_1)/Zp_1 \cdot N$ (24)

$m^2 = - (Zs_2 \cdot Zp_1 - Zs_1 \cdot Zp_2)/Zp_2 \cdot N$

$= - (Zs_3 \cdot Np_2 - Zs_2 \cdot Zp_3)/Zp_2 \cdot N$ (25)

$m^3 = (Zs_2 \cdot Zp_3 - Zs_3 \cdot Zp_2)/Zp_3 \cdot N$

$= - (Zs_3 \cdot Zp_1 - Zs_1 \cdot Zp_3)/Zp_3 \cdot N$ (26)

(m_1, m_2, m_3 are arbitrary integers)

The number of the teeth of each pinion is obtained from the equations (24), (25) and (26). If $Zp_1 = Zp_2 = Zp_3$, the equation is simplified as follows.

$$\left.\begin{aligned} m_1 &= (Zs_1 - Zs_2)/N \\ m_2 &= (Zs_3 - Zs_2)/N \\ m_3 &= (Zs_1 - Zs_3)/N \end{aligned}\right\} \quad (27)$$

It will be seen that the differences among the number of the teeth of the first sun gear 53, the number of the teeth of the second sun gear 54 and the number of the teeth of the third sun gear 55 are integrally multiplied by the number N of the pinion set of the first, second and third pinions 57, 58 and 59. If the module m' of the first pinion 57, the module m'' of the second pinion 58 and the module m''' of the third pinion 59 are $m'' > m''' > m'$ even if the number of the teeth is $Zp_1 = Zp_2 = Zp_3$, the pitch circles become $rp_1 < rp_3 < rp_2$.

Here, Zs_1 is 24, Zp_1, Zp_2 and Zp_3 are 18, respectively, Zs_2 is 15, Zs_3 is 18 and N is 3. If helical gears are used, and the module of each of the teeth Zs_1 and Zp_1 is 1.5, the module of each of the teeth Zs_2 and Zs_2 is 1.75 and the module of each of the teeth Zp_3 and Zs_3 is 1.72 and if rs_1 is 22.8 mm, rp_1 is 17.1 mm, rp_2 is 21.8 mm and rs_2 is 18.1 mm, and rs_3 and rp_3 are 19.95, respectively, the equation (27) becomes

$m_1 = (24 - 15)/3 = 3$

$m_2 = (18 - 15)/3 = 1$

$m_3 = (24 - 18)/3 = 2$

Consequently, the three pinion sets, in which the first, second and third pinions of 57, 58 and 59 each group are arranged at the same phase with each other, can be equiangularly disposed.

Thus, the balance of mass among the three pinions are ensured during the operation, thereby improving durability of bearings and reducing noises and vibration of the central differential 50.

Further, since the three pinions of each group are in the same phase and a plurality of the pinion sets are equiangularly disposed by determining the number of the teeth and measurements of the gears, assembling and workability of the planetary gear device are improved. It is possible to simplify manufacturing of the gears and reduce parts of the pinions.

Since measurements of gears between the first sun gear and the pinion and between the second sun gear and the pinion and between the third sun gear and the pinion can be preferably changed, wide controlling of the torque split can be performed. Thus, operability and driveability of the vehicle are accurately and properly controlled, thereby improving efficiencies thereof.

The present invention may be adapted to a vehicle with a manual transmission or a continuously variable belt-drive automatic transmission, or to a vehicle with a laterally mounted engine.

From the foregoing it will be understood that the present invention provides a torque split control system for a four-wheel drive vehicle where at least two standard torque split ratios are obtained, thereby providing stability of the vehicle, good cornering performance, handling and steerability in accordance with the road conditions and driving conditions so that the driving performance is improved. Since the central differential comprises a complex planetary gear device, a sufficient torque is distributed to the rear wheels.

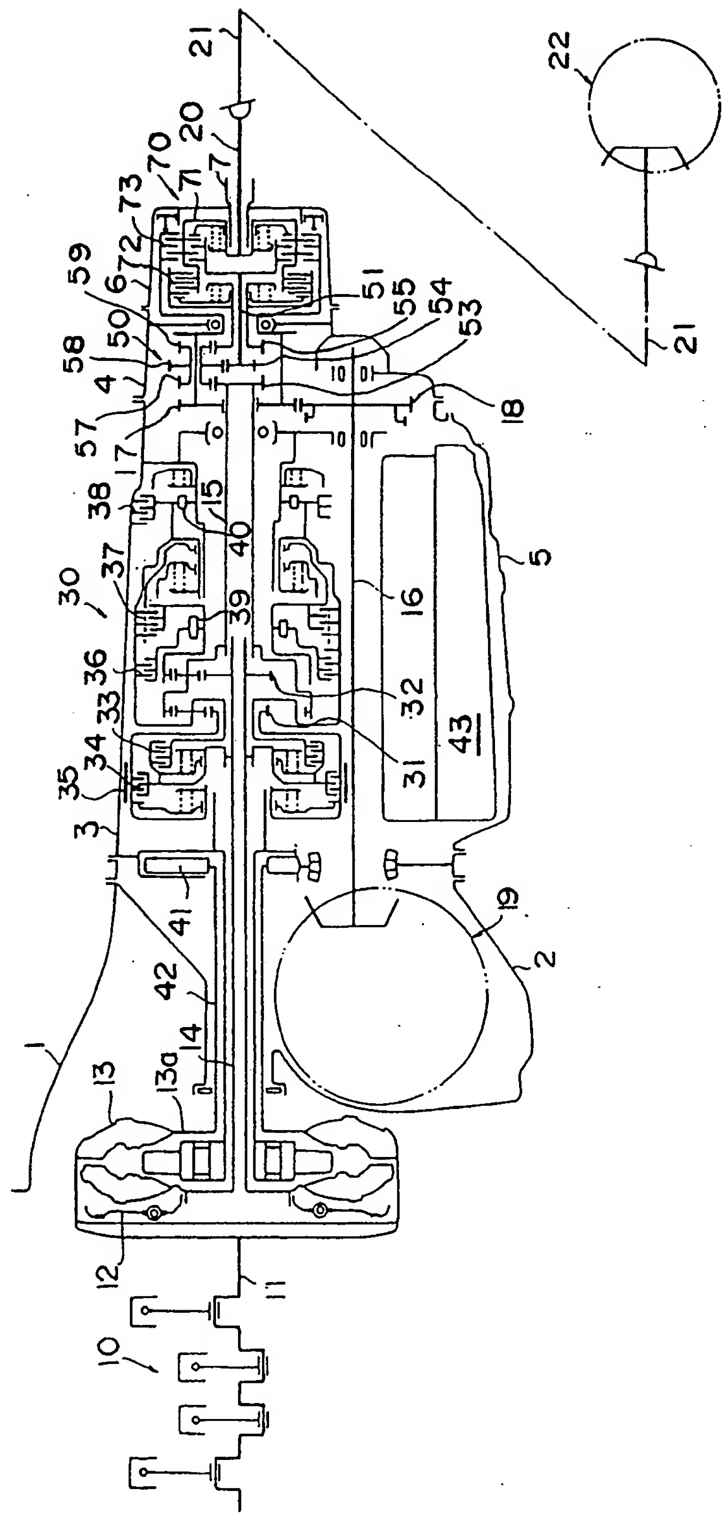
While the presently preferred embodiments of the present invention have been shown and described, it is to be understood that these disclosures are for the purpose of illustration and that various changes and

modifications may be made without departing from the scope of the invention as set forth in the appended claims.

Claims

1. A system for controlling torque distribution in a four-wheel drive vehicle having a transmission (30) and power trains for transmitting the torques to front and rear sets of drive wheels of the vehicle; comprising, a central differential (50) having an input sun gear (53) coupled to an output shaft (15) of said transmission (30), a plurality of output sun gears (54, 55) coaxial with said input sun gear (53), a carrier (61) coupled to one set of the driving wheels of said vehicle, a pinion member comprising a plurality of planetary pinions (57 to 59) directly and coaxialy coupled to each other and each engaged with one of the sun gears (53 to 55) to form ah input planetary gear set and a plurality of output planetary gear sets, each of the output planetary gear sets having a drive ratio different to any other output planetary gear set, selecting means for engaging one of said output sun gears (54, 55) to an output member (20) coupled to another set of the drive wheels; and
a control unit (90) for operating said selecting means in accordance with the driving conditions of the vehicle.
2. The system according to claim 1, wherein
said selecting means comprises fluid operated friction clutches (71, 72).
3. The system according to claim 1, wherein the drive ratio of the input planetary gear set is equal to the drive ratio of one of the output planetary gear sets.
4. The system according to one of claims 1 to 3 comprising:
a fluid operated multiple-disk clutch (73) to restrict the differential operation.
5. The system according to any of the preceding claims, wherein
said planetary pinions are arranged at the same phase; and
a plurality of said pinion members are equiangularly disposed around said sun gears.
6. The system according to any one of the preceding claims comprising:
means for rendering said central differential operative during a period of time when said selecting means operates to select one of said output sun gears (54, 55), thereby preventing a decrease of the torque during the period.

FIG. 1


21
22
20
7
70
71
73
72
6
59
50
58
4
57
17
51
55
54
53
18
38
15
40
37
30
39
16
5
36
33
32
34
35
31
43
3
41
19
42
2
14
1
13
13a
12
11
10

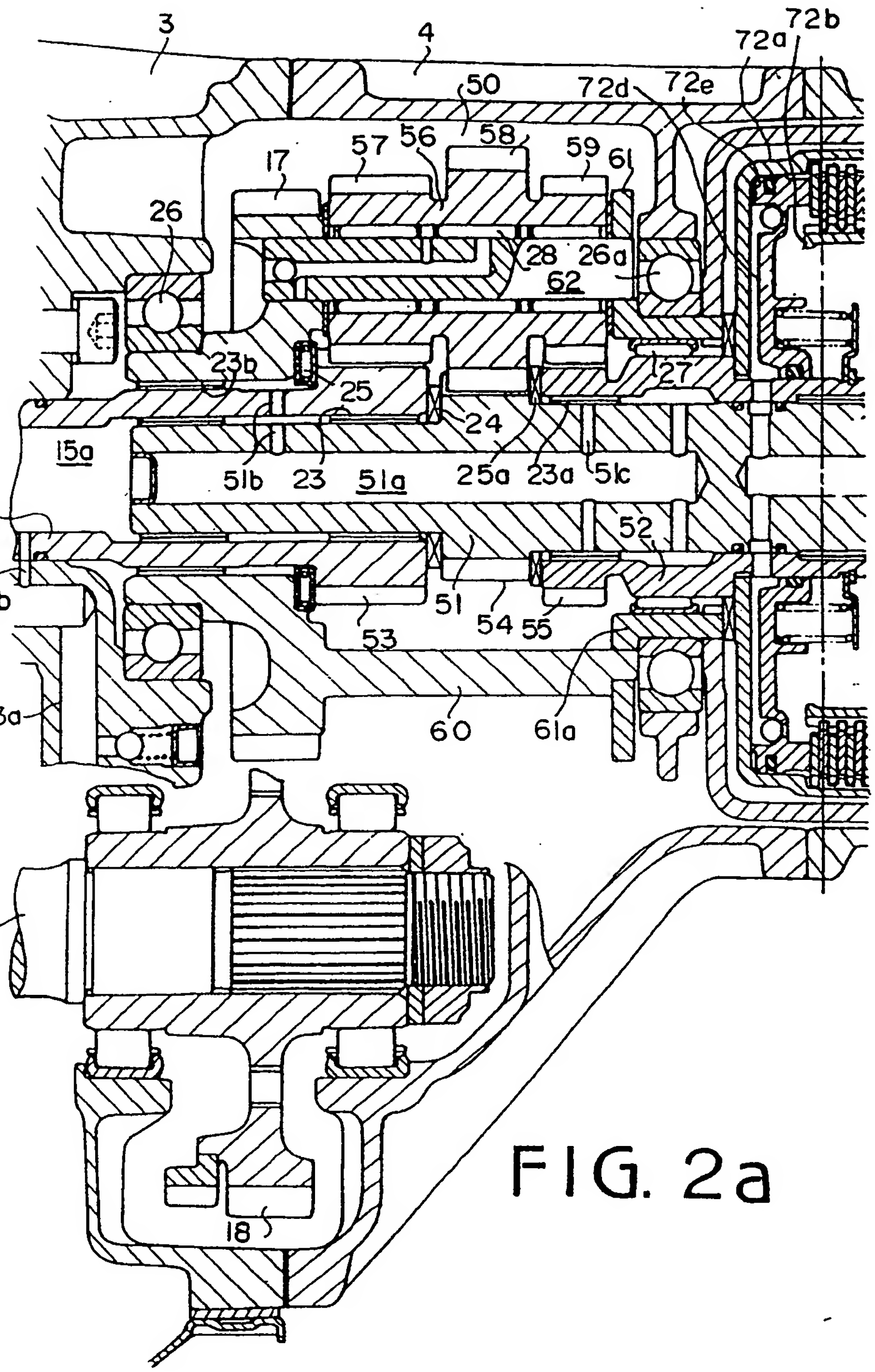

3
4
50
72d
72e
72a
72b
57
56
58
59
61
17
26
28
26a
62
23b
25
27
24
15a
51b
23
51a
25a
23a
51c
15
52
15b
51
54
55
53
3a
60
61a
16
18

FIG. 2a

FIG. 2b

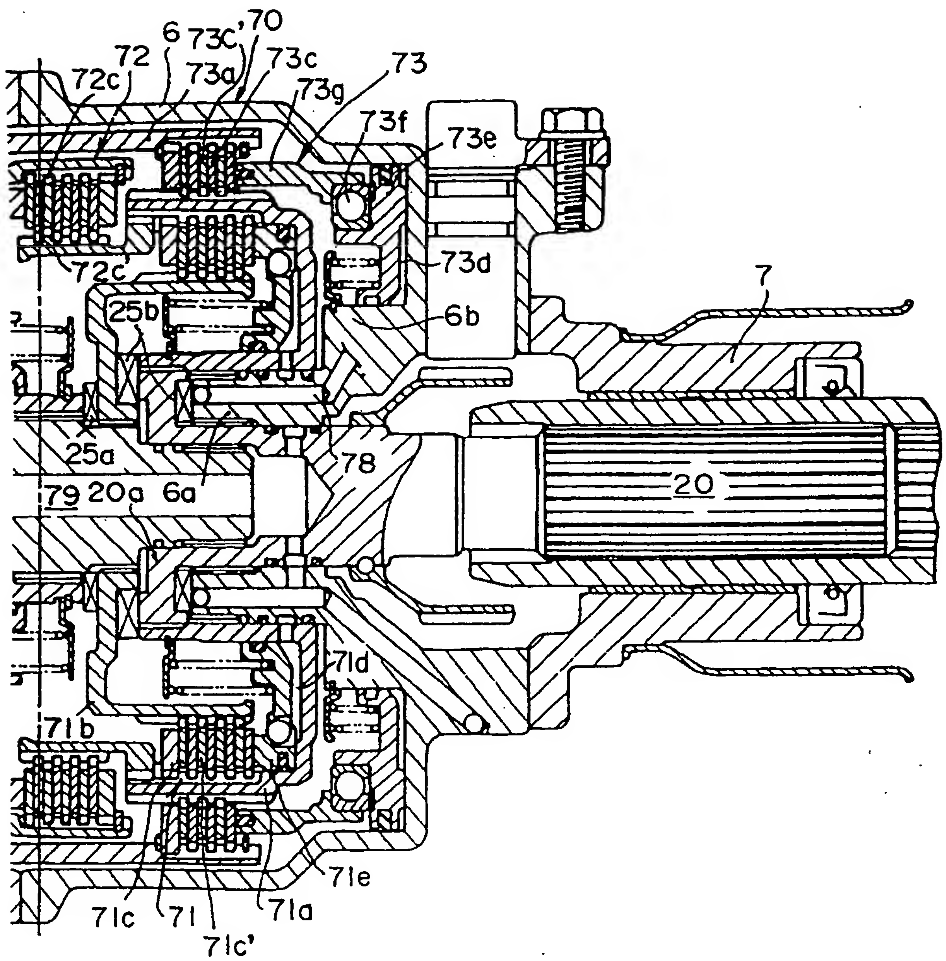
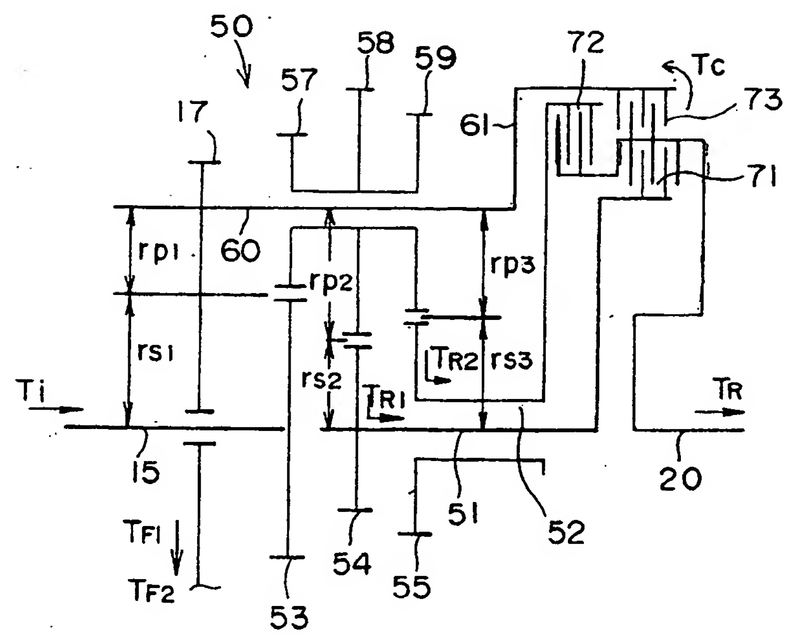

6
73C'
70
72
73c
73
72c
73a
73g
73f
73e
72c'
73d
7
25b
6b
25a
78
79
20a
6a
20
71d
71b
71e
71c
71
71a
71c'

FIG. 3



50
57
58
59
72
Tc
73
17
61
71
rp1
60
rp2
rp3
rs1
TR2
rs3
Ti
rs2
TR1
TR
15
51
20
TF1
52
TF2
53
54
55

FIG. 4a

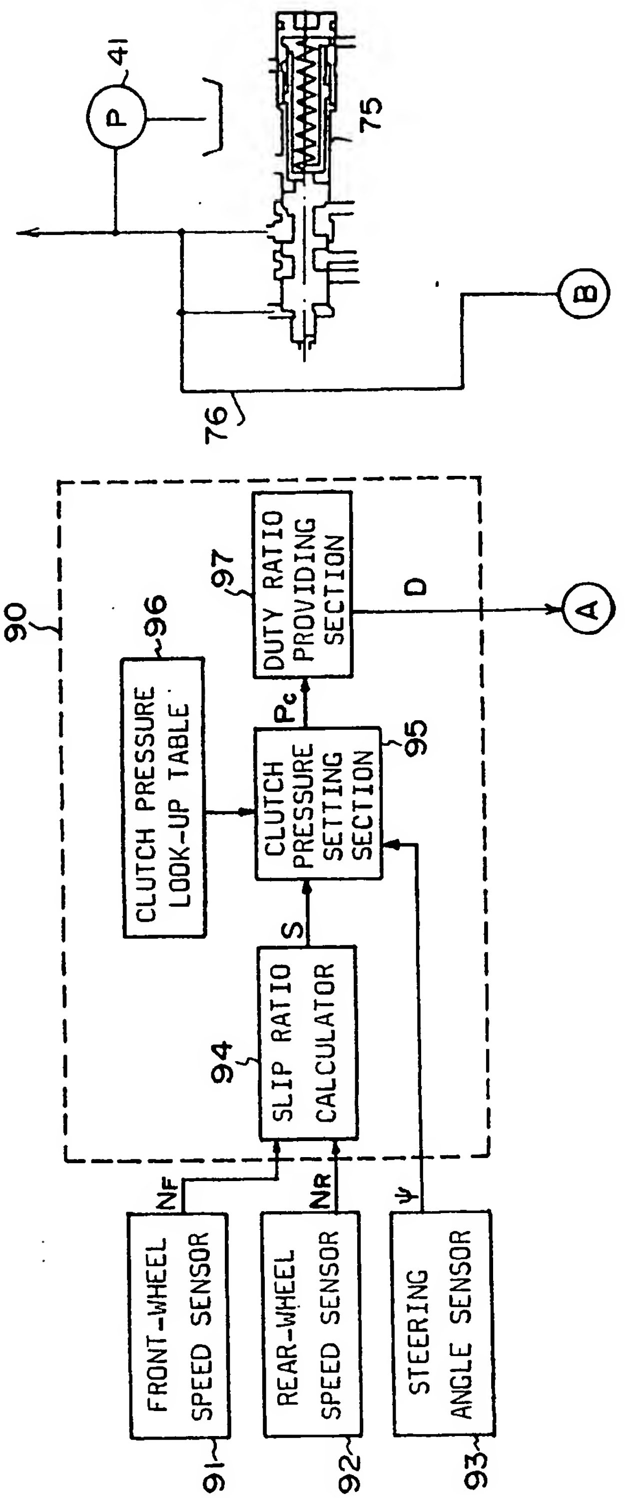

41
P
75
B
76
90
96
CLUTCH PRESSURE LOOK-UP TABLE
97
DUTY RATIO PROVIDING SECTION
D
A
Pc
95
CLUTCH PRESSURE SETTING SECTION
S
94
SLIP RATIO CALCULATOR
NF
NR
ψ
FRONT-WHEEL SPEED SENSOR
REAR-WHEEL SPEED SENSOR
STEERING ANGLE SENSOR
91
92
93

FIG. 4b

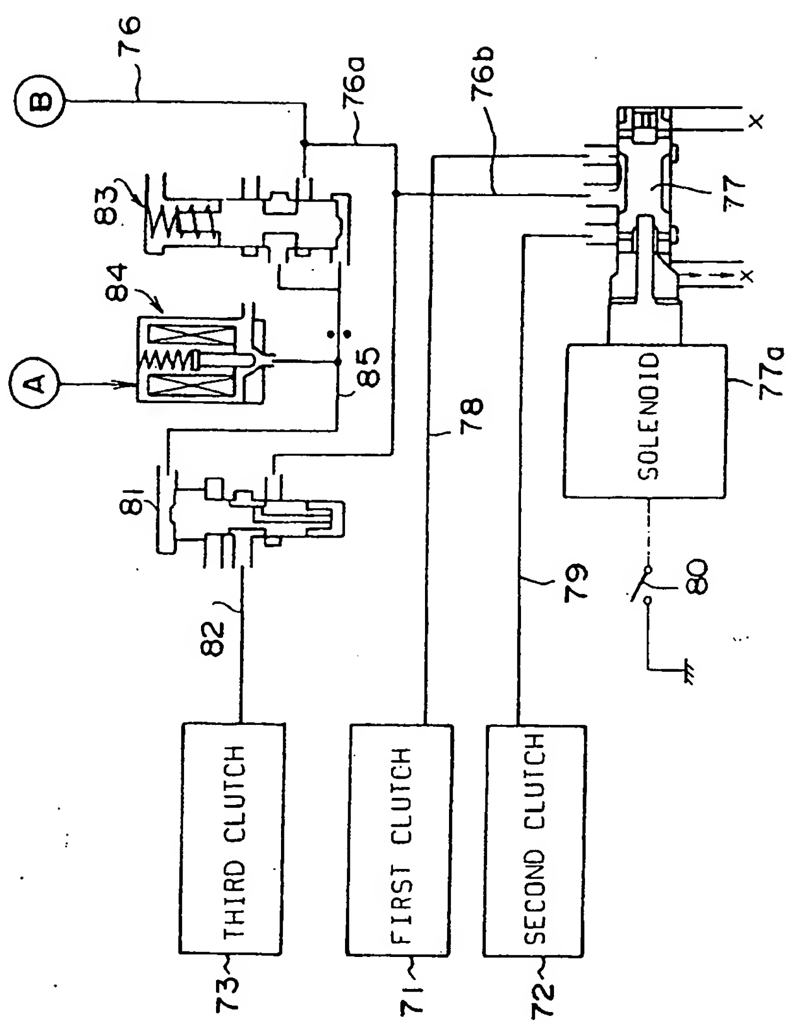

A
B
76
76a
76b
83
84
85
81
82
78
79
77
77a
80
X
SOLENOID
THIRD CLUTCH
FIRST CLUTCH
SECOND CLUTCH
73
71
72

FIG. 5a

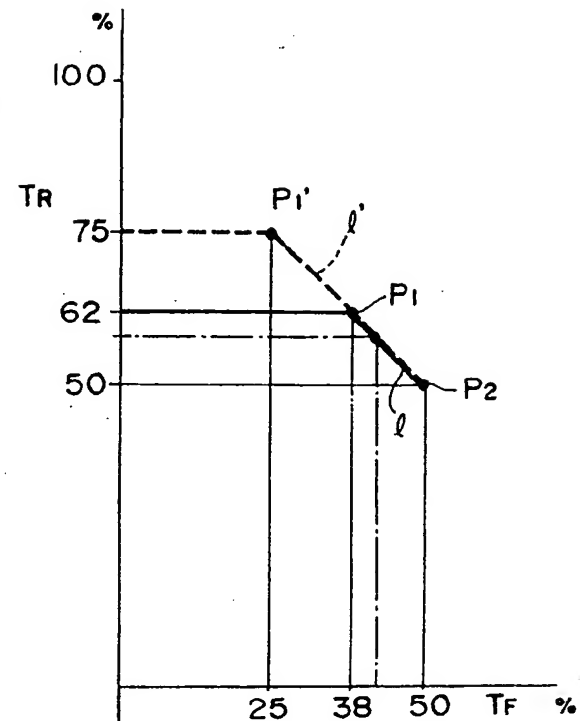

%
100
TR
75
62
50
P1'
ℓ'
P1
P2
ℓ
25
38
50
TF
%

FIG. 5b

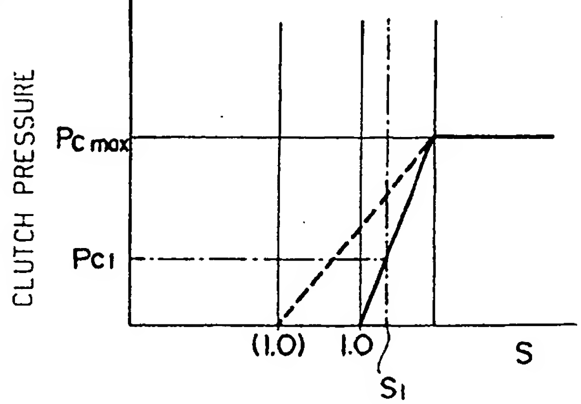

CLUTCH PRESSURE
Pc max
Pc1
(1.0)
1.0
S1
S

FIG. 6

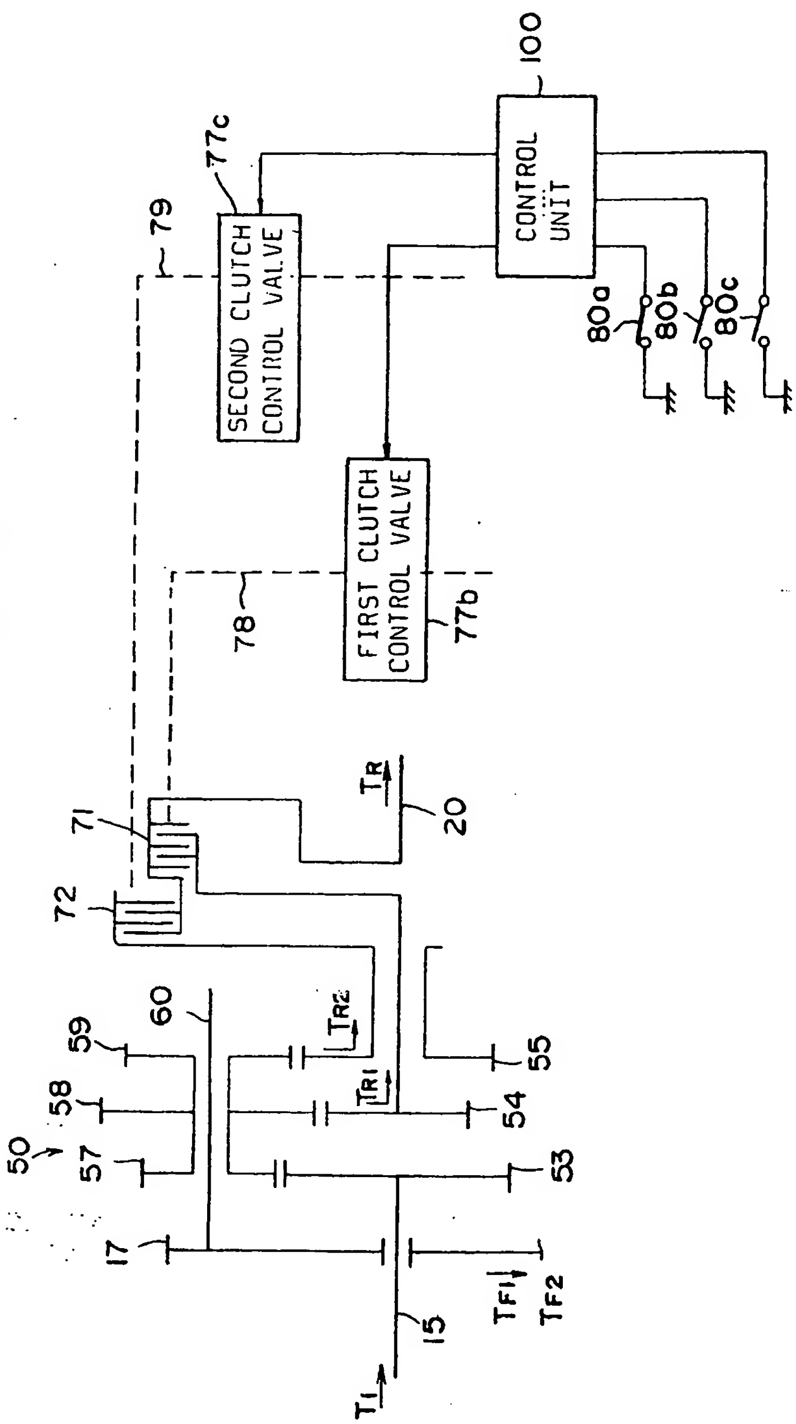

SECOND CLUTCH CONTROL VALVE
FIRST CLUTCH CONTROL VALVE
CONTROL UNIT
100
77c
77b
79
78
80a
80b
80c
71
72
20
T_R
60
T_{R2}
T_{R1}
59
58
57
50
55
54
53
17
15
T_I
T_{F1}
T_{F2}

FIG. 7a

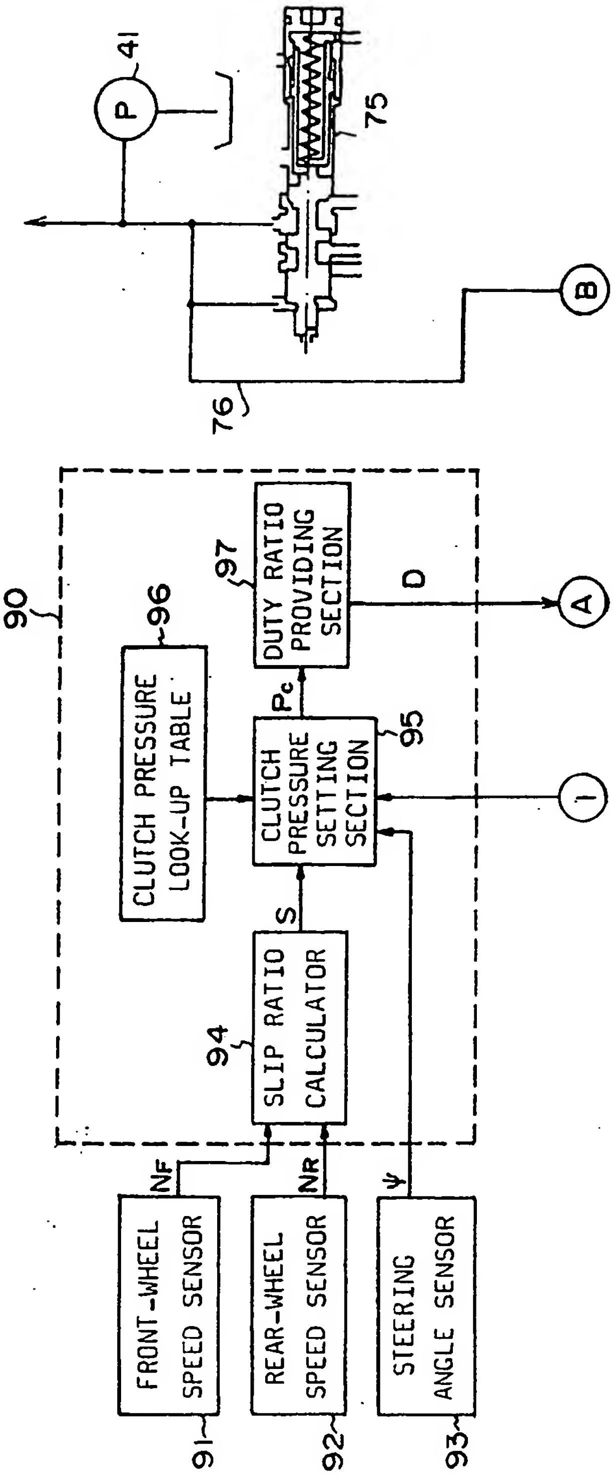
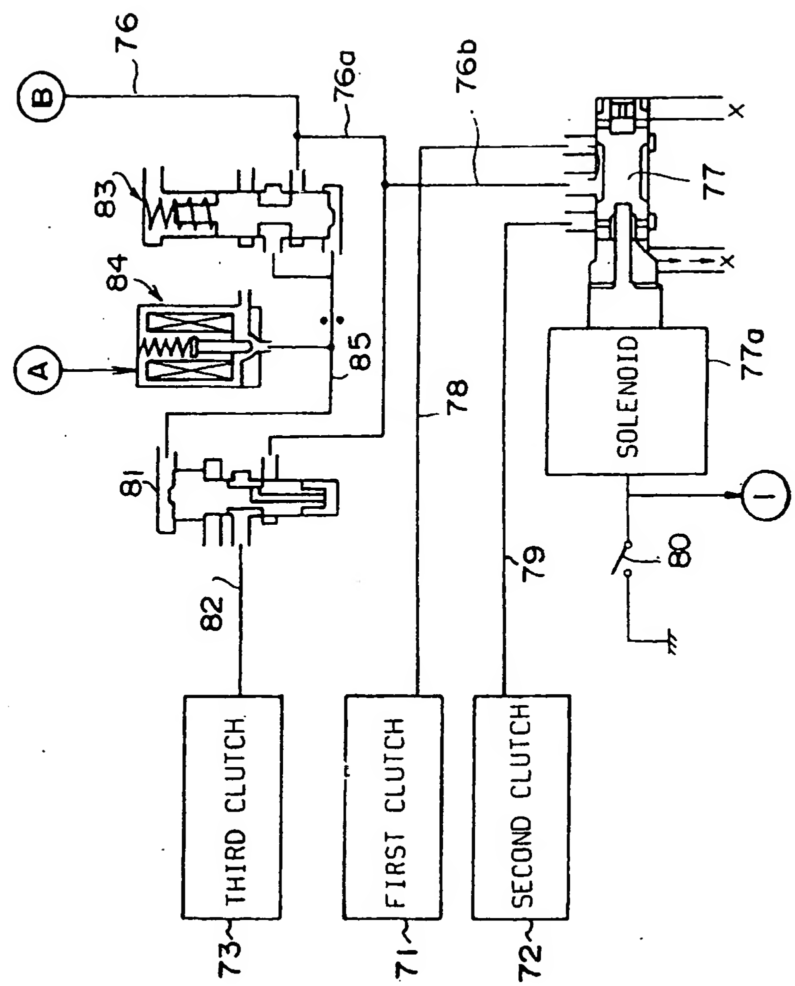

41
P
75
76
B
90
96
CLUTCH PRESSURE LOOK-UP TABLE
97
DUTY RATIO PROVIDING SECTION
D
A
Pc
95
CLUTCH PRESSURE SETTING SECTION
I
S
94
SLIP RATIO CALCULATOR
NF
NR
ψ
FRONT-WHEEL SPEED SENSOR
REAR-WHEEL SPEED SENSOR
STEERING ANGLE SENSOR
91
92
93

FIG. 7b



B
76
76a
76b
83
84
A
85
81
82
78
79
77
77a
X
SOLENOID
I
80
THIRD CLUTCH
FIRST CLUTCH
SECOND CLUTCH
73
71
72

FIG. 8a

FIG. 8b

FIG. 8c

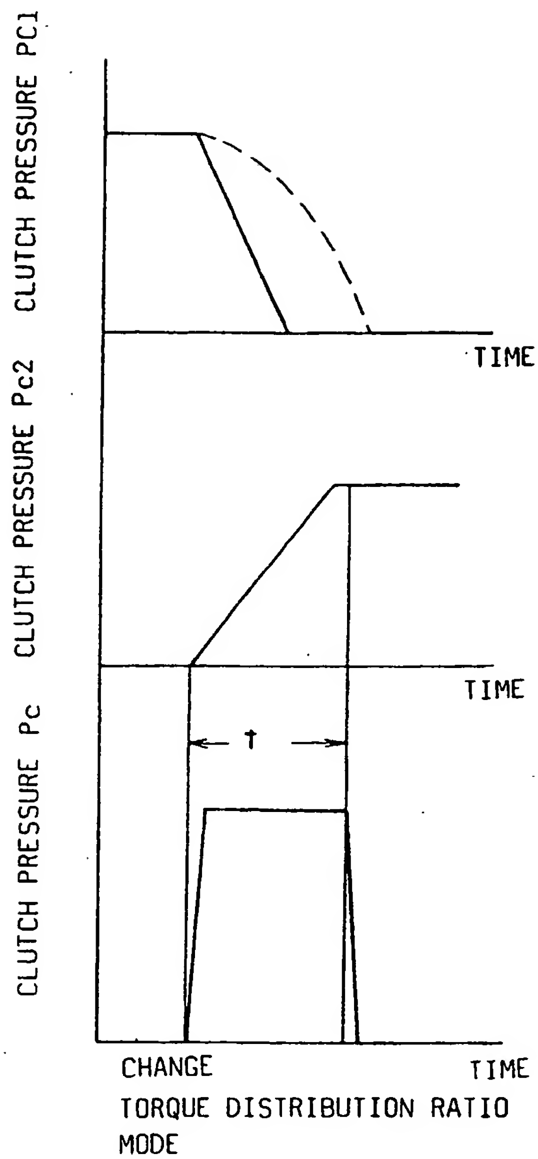


FIG. 9

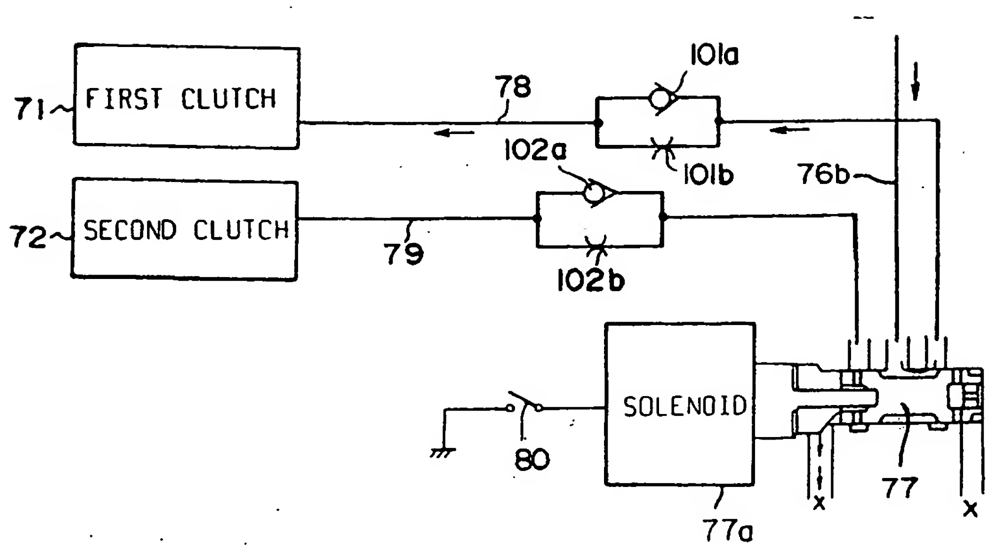

71
FIRST CLUTCH
78
101a
101b
76b
102a
72
SECOND CLUTCH
79
102b
SOLENOID
80
77
77a
X
X

FIG. 10

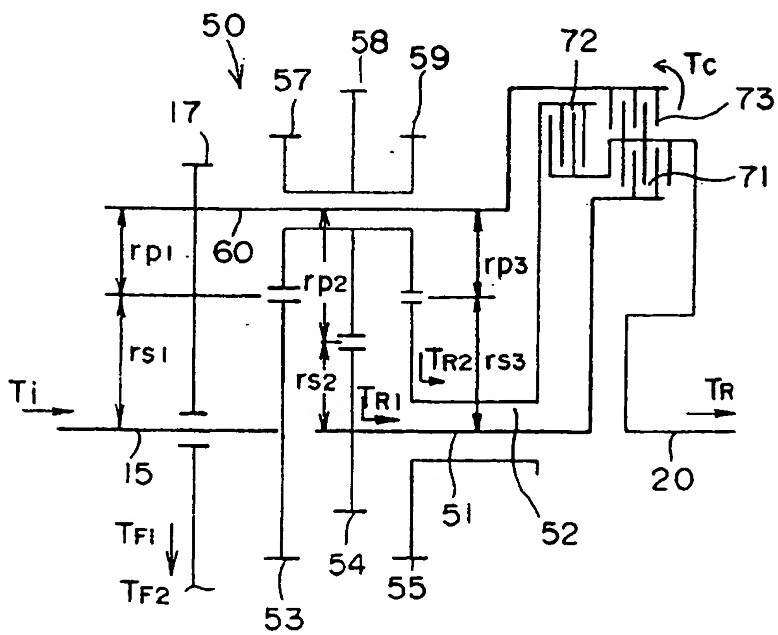

50
58
57
59
72
Tc
73
17
71
rp1
60
rp3
rp2
rs1
TR2
rs3
rs2
Ti
TR1
TR
15
20
51
52
TF1
54
TF2
55
53

FIG. 11a

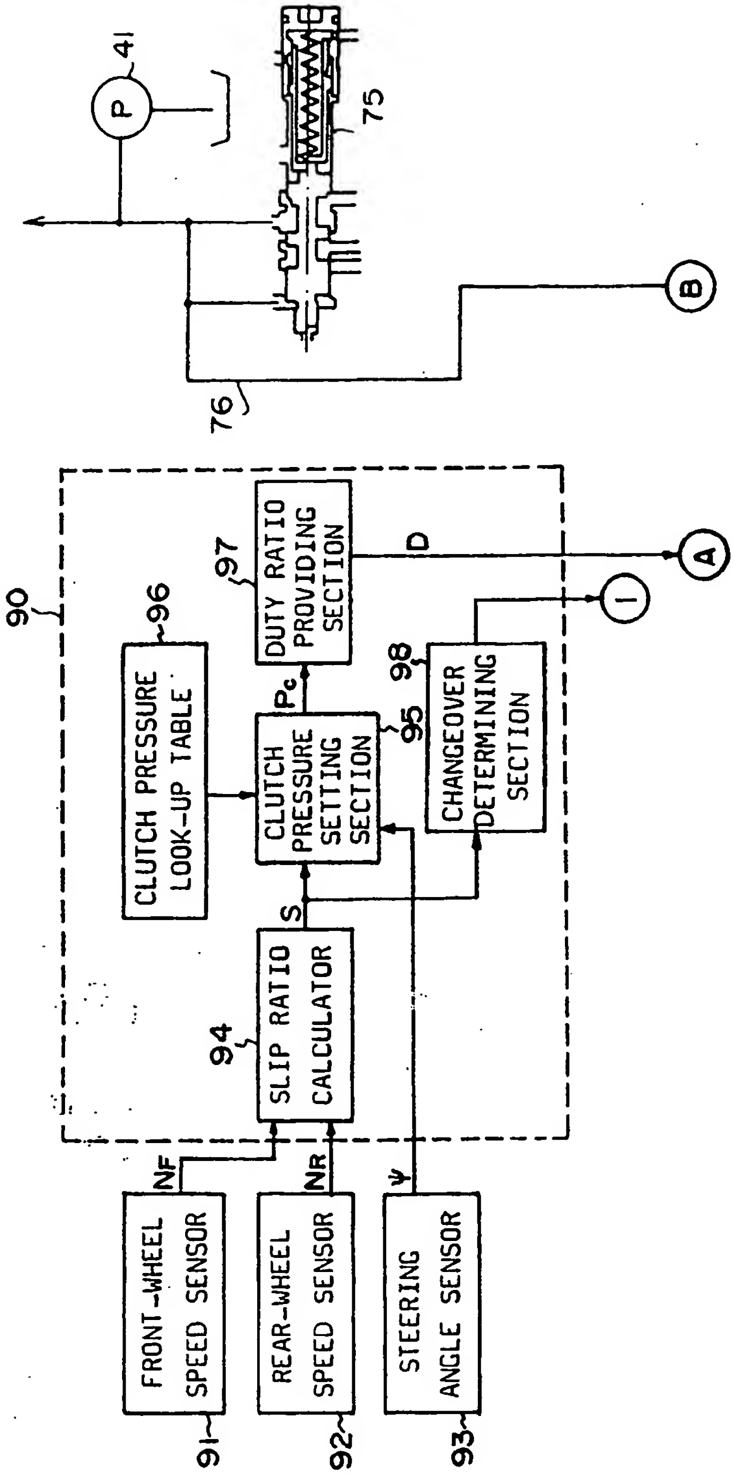

41
P
75
76
B
90
96
CLUTCH PRESSURE LOOK-UP TABLE
97
DUTY RATIO PROVIDING SECTION
D
A
I
Pc
95
CLUTCH PRESSURE SETTING SECTION
98
CHANGEOVER DETERMINING SECTION
S
94
SLIP RATIO CALCULATOR
NF
NR
ψ
FRONT-WHEEL SPEED SENSOR
91
REAR-WHEEL SPEED SENSOR
92
STEERING ANGLE SENSOR
93

FIG. 11b

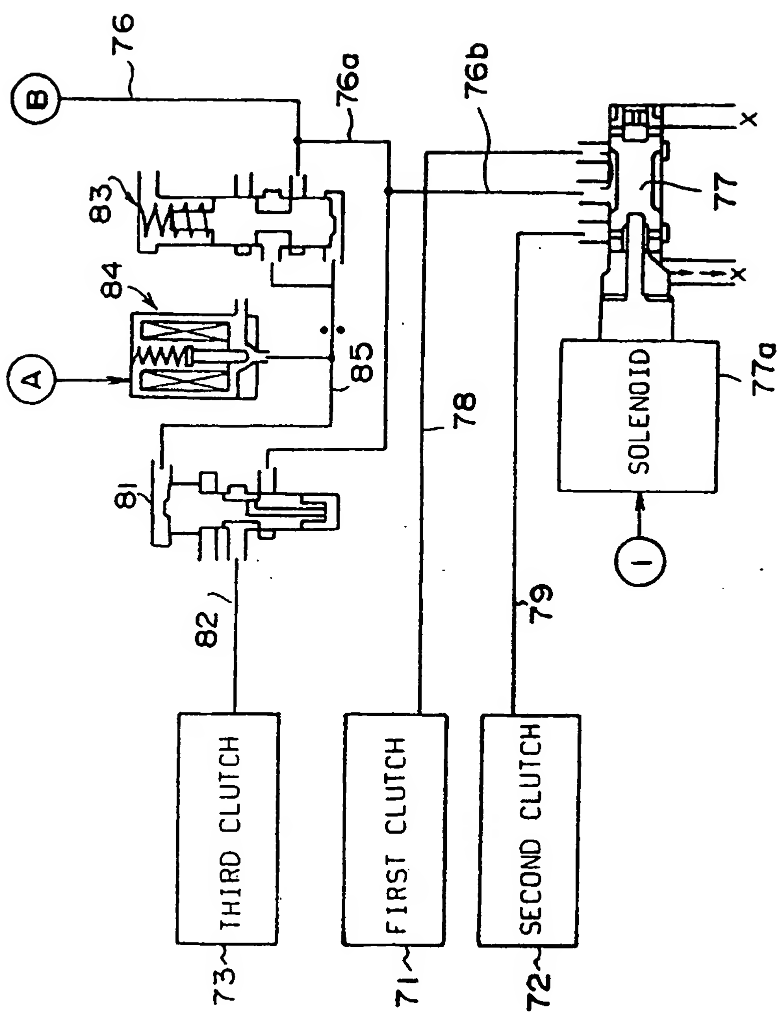

B
A
I
76
76a
76b
77
77a
78
79
81
82
83
84
85
X
X
SOLENOID
THIRD CLUTCH
FIRST CLUTCH
SECOND CLUTCH
73
71
72

FIG. 12a

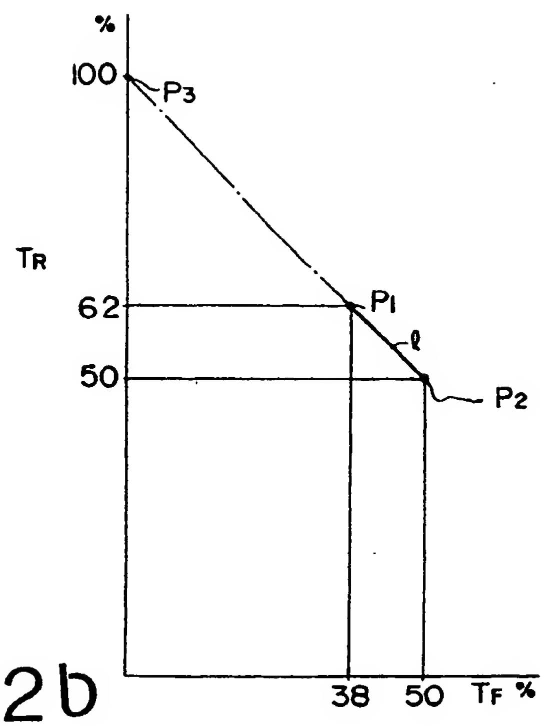


FIG. 12b

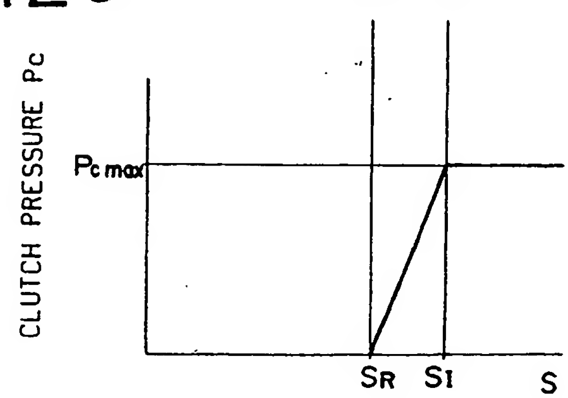


FIG. 13a

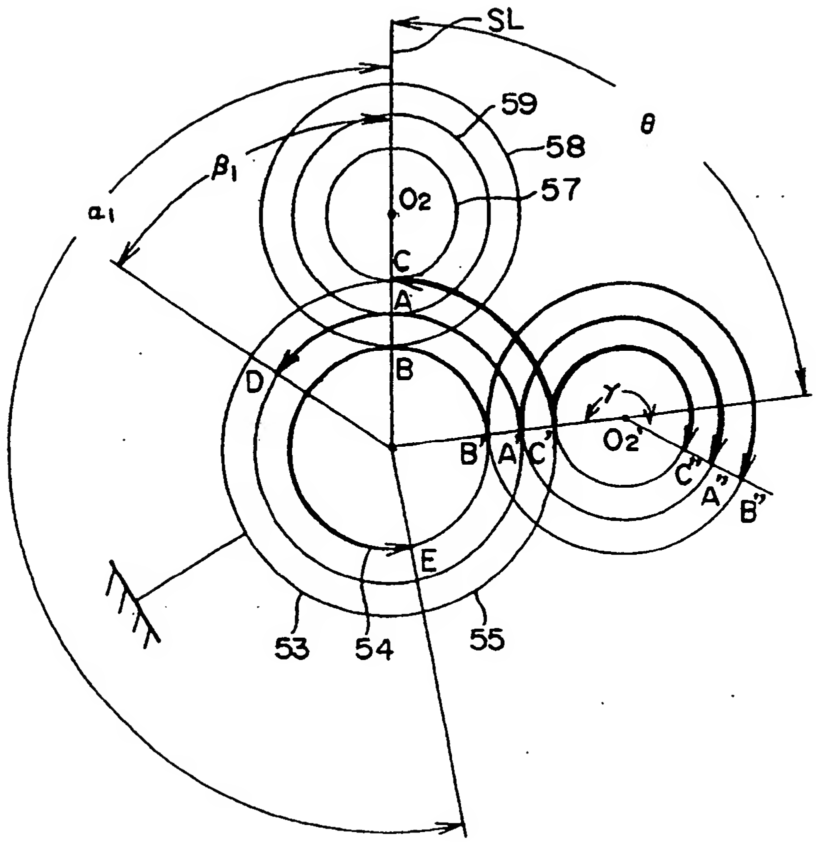


FIG. 13b

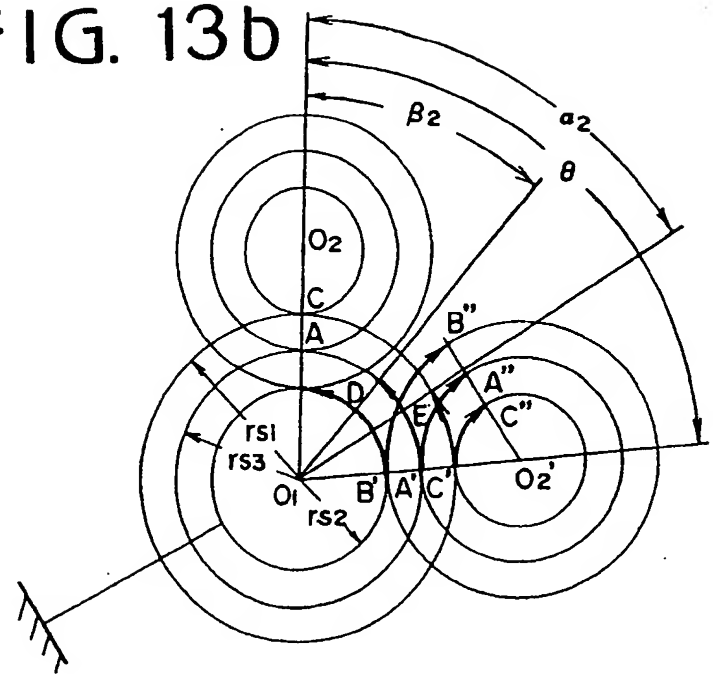
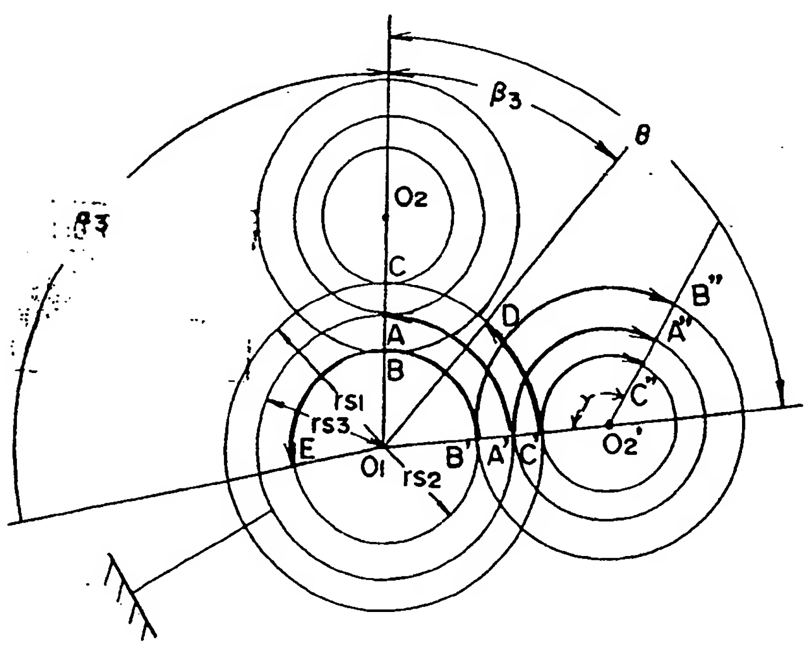


FIG. 13c



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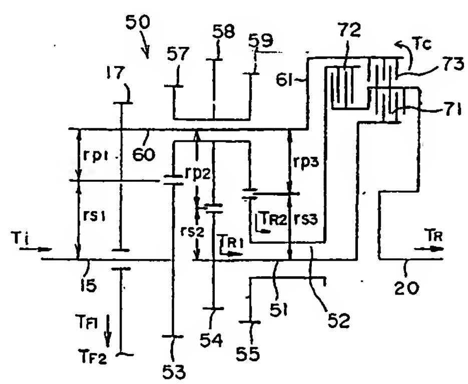
(54) **A system for controlling torque distribution in a four wheel drive vehicle.**

(57) A central differential (50) for splitting output torque of a transmission is formed by a complex planetary device. The planetary device comprises an input sun gear (53) operatively connected with an output shaft (15) of the transmission, a plurality of output sun gears (54, 55), a carrier (61), a pinion member (56) comprising a plurality of planetary pinions (57, 58, 59) integral with each other and rotatably supported on the carrier. Fluid operated friction clutches (71, 72, 73) are provided for selectively connecting one of the output sun gears to an output member (20) of the central differential.

FIG. 3



EP 0 424 054 A3

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EUROPEAN SEARCH REPORT

Application Number

EP 90 31 1240

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int. Cl.5)
A	DE-A-3 714 334 (MAZDA MOTOR CORP.) * the whole document *	1-4	B 60 K 17/346 B 60 K 23/08
A	DE-A-3 706 506 (FUJI JUKOGYO K.K.) * column 3, line 5 - column 4, line 54; figure 1 *	1-4	
A	PATENT ABSTRACTS OF JAPAN vol. 10, no. 227 (M-505)(2283) 07 August 1986, & JP-A-61 62641 (TOCHIGI FUJI IND CO LTD) 31 March 1986, * the whole document *	1	
D,A	PATENT ABSTRACTS OF JAPAN vol. 12, no. 445 (M-767)(3292) 22 November 1988, & JP-A-63 176728 (AISIN WARNER LTD) 21 July 1988, * the whole document *	1	
			TECHNICAL FIELDS SEARCHED (Int. Cl.5)
			B 60 K F 16 H

The present search report has been drawn up for all claims

Place of search	Date of completion of search	Examiner
The Hague	28 August 91	TOPP-BORN S.

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